TECHNICAL NOTES

NATIONAL ADVISORY COMMITTEE FOR AERONAUTICS

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A FULL-SCALE INVESTIGATION OF THE EFFECT OF SEVERAL FACTORS ON THE SHIMMY OF CASTERING WHEELS

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SUMMARY

A full-scale investigation has been conducted to determine the effect of various factors on the shimmy of castering wheels. The factors considered were the geometric arrangement, the tire types, the variations of load, the spindle moment of inertia, and the tire inflation. A comparison of the results of the present investigation with those calculated from existing theory was made. The constants needed in the calculations to determine the damping required for a castering wheel were measured.

The results indicate that solid friction appears to be impracticable as the sole damping agent for castering nose wheels on large airplanes. Also it was concluded that the existing theory is adequate for calculating the damping required to prevent shimmy. The caster angle and the spindle moment of inertia were found to influence the solid friction required to prevent shimmy. The effect of variations in the type and the pressure of the tire was insignificant.

INTRODUCTION

The National Advisory Committee for Aeronautics has conducted a full-scale investigation of the shimmy (spindle oscillations due to dynamic instability) of castering wheels. In addition to the determination of the effect of the physical properties of a castering wheel on the tendency to shimmy, data were obtained by which the applicability of a previous theoretical investigation (reference 1) could be examined.

The investigation included the following factors that must be considered in the design of a castering-wheel sys-

tem: the geometry of the caster system (caster angle, fork offset, and caster length); the load; the spindle moment of inertia, the wheel moment of inertia; and the tire inflation. These variables were examined on the basis of their effect on the tendency to shimmy while the test wheel was towed over a uniform bump. This testing does not contribute data amenable to an analysis leading to quantitative design information that would be applicable to all sizes and types of castering-wheel systems; it is believed, however, that a qualitative analysis is valid and is useful for systems where solid friction is used.

Certain tire constants described in reference 1 have been evaluated to allow a comparison between the test values and the theoretical values of solid friction required to prevent shimmy.

APPARATUS

Shimmy Equipment

The castering wheels and tires were mounted on a cart, which was drawn by a tow car. The characteristics of the five tires tested are shown in table I. The tow car used was a standard make of automobile capable of rapid acceleration up to 50 miles per hour in second gear. The assembled apparatus is shown in figure 1.

The design of the tow cart provided for several imposed test requirements; namely, the variation of load, the adjustment of the caster angle and the fork offset, the application and the measurement of solid friction, and the interchangeability of tires. The cart was attached by two bolts to the tow-car frame in a manner that allowed the cart to bounce but restricted the sidewise motion of the cart to the movement of the car or the car springs. The load was varied by changing the number of sand bags in the weight compartments provided in the cart. A frame was attached to the cart proper by means of two pivots and an adjusting screw that provided adjustment of the caster angle. The variation of the fork offset was obtained by an adjustment of the fork angle. (See fig. 2.) The upper end of the spindle was attached by a removable pin to a brake drum, which in turn was encircled by a spring-loaded brake band providing adjustable solid fric-

tion. Figure 3 shows the upper assembly of the spindle and the brake drum.

The bump used was made of half-round pine with a 3-inch base, 24 inches long, and had a plate attached to the center so that the bump could be secure to the street. The axis of the bump made an angle of 45° with the path of the towed wheel.

Hammond Y-1 Airplane

The airplane used for these tests was a Hammon Y-1, low-wing, two-place, pusher monoplane with a tricycle landing gear. The gross weight of the airplane was 2,200 pounds. The caster angle of the nose wheel was 20° with no fork offset. The bump previously described was also used for these tests.

Equipment for Determination of Tire Constants

In order to determine three of the tire constants, the spindle frame from the tow cart and the wheels were attached to the N.A.C.A. tank towing carriage (reference 2) as shown in figure 4. Satisfactory control of the weight on the tires was provided by this arrangement, but the inability of the carriage to maintain a slow, uniform velocity made it difficult to get an accurate determination of the quantities used to evaluate the tire constants. The data used to calculate the fourth constant were obtained with the tire resting on a wire-supported platform as shown in figure 5. This arrangement provided a means of measuring the side deflection of the tire for a known side force. The deflections were measured with a dial indicator mounted on the platform.

METHOD AND PROCEDURE

The testing program, which was intended to determine the effect of variables not covered by the theory as well as to test theoretical results, was divided into three phases: the full-scale shimmy tests on the tow cart to evaluate the significance of various test variables; the full-scale shimmy investigation of the Hammond Y-1 airplane with tricycle landing gear; and the determination of tire constants.

Full-Scale Shimmy Tests with Tow Cart

Prior to actual shimmy tests, it was necessary to determine the tire rolling radius under conditions of design pressure and maximum allowable static load. This rolling radius was maintained as nearly constant as possible in each series of tests except during the determinations of the effect of underinflation or overinflation. The testing procedure consisted in first adjusting the caster angle, the fork offset, the load, the rolling radius, and the applied solid friction to preselected values. The car with the tow cart was then driven at a constant speed along a concrete road so that the wheel struck the bump. If no shimmy occurred, the speed for the succeeding run was increased approximately 5 miles per hour until a speed of about 50 miles per hour was reached. When shimmy occurred, the friction was increased, determined, and the run was repeated. In this way, the friction required to prevent shimmy at 50 miles per hour for each of the test conditions was established. These observed values of friction were corrected by adding the amount of friction torque in the spindle bearings at various values of load and under different geometric conditions.

The cart structure limited the range of caster angles for most of the tests to between 10° and −15°. The values of fork offset used were 1, 1.9, and 2.8 inches. The loads used were 480 and 640 pounds. When shimmy occurred with larger loads, structural failure usually resulted.

The variations in spindle moment of inertia and wheel moment of inertia were obtained by the addition of lead weight to the part in question.

Full-Scale Shimmy Tests with Hammond Y-1 Airplane

Tests were made with the Hammond Y-1 airplane to determine the solid friction required to prevent shimmy of the nose wheel when the airplane was taxied on a concrete runway over the bump used in the shimmy tests. The speed and the load were varied, the load variation being accomplished by a change in the elevator deflection. The nose-wheel load was measured by photographically recording the amount of compression of the nose-wheel oleo strut. The solid friction was varied by inserting or removing shims

of various size in a clamping block on the oleo strut. The amount of friction was determined by lifting the wheel clear of the ground on a sling and determining the torque necessary to turn the spindle.

The amount of friction that was employed may be described as follows: (1) Steering was extremely difficult; (2) steering was possible but stiff; (3) steering was reasonably free.

Determination of Tire Constants

In order to compare the experimentally determined values of friction with those predicted on the basis of existing theory, the constants described in reference 1 were determined for three tires. The difficulties encountered in these measurements made it inadvisable to conduct tests of the other two types of tire.

The determination of E_λ, the lateral force exerted by the tire per foot deflection, required the application of a side force F to the bottom of the tire and the measurement of the lateral deflection of the tire λ produced with respect to the plane of the wheel. The side force was measured with a spring balance, the wheel having been suspended on a sling and a platform as shown in figure 5. The deflection of the bottom of the tire with respect to the plane of the wheel was measured with a dial indicator mounted on the platform. The values of F and λ were measured at different loads, the rolling radius being maintained constant, and resulted in

$$E_\lambda = F/\lambda \text{ lb/ft}$$

The constants K_1, C_1, and C_2 were determined with the wheel attached to the tank towing carriage as shown in figure 4.

The constant K_1 was determined by the measurement of the interval of kinematic shimmy S and substitution in the equation

$$K_1 = \frac{(2\pi)^2}{S^2} \text{ ft}^{-2}$$

This equation and the ones that follow were determined

from equations taken from reference 1. The determination of S required observation of the wave length of an oscillation occurring when the spindle was given an initial rotation and the wheel was slowly rolled ahead.

The determination of the constant C_1 required a measurement of the torque T necessary to hold the plane of the wheel at a constant known angle θ, the angle between the plane of the wheel and the direction of motion, while the wheel moved slowly ahead.

A basis for the calculation of C_1 is given by the equation

$$T = C_1 I_W \lambda$$

Since, for steady motion, $\theta = - C_2 \lambda$ by substituting for λ

$$C_1 I_W = - \frac{TC_2}{\theta}$$

where I_W is the spindle moment of inertia.

In order to determine C_2, the wheel was moved forward as slowly as possible and the bottom of the tire was given an initial deflection. The reduction of torque $(T_o - T)$ was observed for a measured distance, $s - s_o$. The angle between the plane of the wheel and the direction of motion was held at zero. The initial deflection with the carriage in motion was obtained by rotating the spindle several degrees, causing the tire to run to one side, and then quickly turning the spindle so that the wheel angle was zero.

Inasmuch as $T = C_1 I_W \lambda$ and $T_o = C_1 I_W \lambda_o$, $T/T_o = \lambda/\lambda_o$ and, from reference 1, $\log_e \lambda/\lambda_o = - C_2(s - s_o)$. By substitution, an equation that provides a basis for the measurement of C_2 can be obtained

$$\log_e T/T_o = -C_2(s - s_o)$$

RESULTS AND DISCUSSION

Full-Scale Shimmy Tests on Tow Cart

The tendency to shimmy as indicated by the solid friction required to prevent shimmy for all the tires tested, except the solid tire,* is plotted against caster angles in figures 6 and 7. Reference to these figures shows that the tendency to shimmy varies only a small amount with the different tires and is insufficient to influence the selection of any particular type when solid friction is employed. The values of solid friction shown should be considered only as minimum values applicable to similar conditions. It should be noted that, whereas a more severe bump might be encountered, it would be unlikely on concrete or on surfaces having an equally high coefficient of friction. (See reference 3.) As discussed later, the values obtained with the tow cart when applied to the Hammond Y-1 airplane were sufficient to prevent shimmy when hitting the half-round bump and other obstacles encountered on and off the runway. For the tires tested, the tendency to shimmy was less at zero or small negative caster angles than at the larger positive angles, which was also indicated from the development of the theory in reference 1.

The results of the effect of spindle moment of inertai I_W are plotted in figure 8. For these tests, the spindle moment of inertia was increased from 0.08 slug-feet^2 to 0.10 slug-feet^2 during the tests of the low-pressure tire.

The effect of load is shown in figures 9 and 10 and indicates that the solid friction required to prevent shimmy is directly proportional to the load. This result is in agreement with the theoretical predictions for solid friction requirements.

The variations in caster angle and the value of zero fork offset represented in figure 10 were chosen to represent conditions existing with the Hammond Y-1 airplane. The tests were conducted with the tow cart and were checked with the airplane. The results show that, with large nose-wheel loads, the particular installation on this airplane required so much solid friction on the nose wheel that steering is rendered very difficult.

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*Owing to the large values of friction damping required and to the severity of shimmies and shocks encountered when the solid tire was tested, no data pertinent to the present investigation were obtained with this type of wheel.

Figure 11 indicates that small changes in fork offset may be made to give the desired caster length without materially affecting the tendency to shimmy. This result may not hold true, however, under conditions other than those of the test and should not be given as a general statement. Also, in any nose-wheel design, the possibility of negative caster length due to bumps, mud, etc., must be kept in mind because the resulting instability of the wheel may readily cause a structural failure.

The effect of variation of the wheel moment of inertia about the wheel axis was very small and the variations that did occur may be attributed to the consequent increase in spindle moment of inertia when mass was added to the wheel.

The variations in tire-inflation pressure of ±15 percent resulted in changes in the solid friction required that were too small to be of interest.

Inconsistencies in the results as indicated by the scatter of the points shown on the curves are probably due to errors in determining the friction of the system and in variations of reproduction of test conditions, particularly as regards the manner in which the wheel struck the bump.

Shimmy Tests with Hammond Y-1 Airplane

In order to prevent shimmy in the nose wheel of the Hammond Y-1 airplane when it was taxied over the standard bump at approximately 50 miles per hour and when the maximum load was applied by fully depressing the elevators, it was necessary to have a solid friction of 250 pound-inches. This solid friction allowed the nose wheel to be steered with difficulty at low speeds. Reducing the solid friction to 190 pound-inches made steering easier but also allowed a shimmy to occur at loads above 750 pounds. Solid friction of 300 pound-inches made steering of the nose wheel nearly impossible and normal turns could be accomplished only with excessive braking of the rear wheels.

The results of tests made with the tow cart are plotted in figure 10 and show that, for conditions used on the Hammond Y-1 airplane with a nose-wheel loading of 750 pounds, solid friction of approximately 240 pound-inches was required to prevent shimmy. This value of solid fric-

tion is in agreement with the preceding test and corresponds to a solid friction value of $T_{max} = 0.05\ r\mu W$.

It is imperative in any system of solid friction that there be no free motion in the spindle system. Any motions not resisted by the solid-friction forces may allow an incipient shimmy to occur that would otherwise be damped.

Tire Constants

Where no tests are available, the theoretical values for solid and viscous friction using the tire constants are the only known quantitative values available. The values for the various tire constants measured are given in table II, and values of E_λ for various loads are plotted in figure 12.

Substitution of these constants in the equations developed for solid friction gives the following values for solid friction required:

Airwheel . . . . $T_{max} = 0.06\ r\mu W$ lb-ft

Low-pressure tire . . $T_{max} = 0.03$ to $0.06\ r\mu W$ lb-ft

Streamline tire . . . $T_{max} = 0.02\ r\mu W$ lb-ft

where r is the tire radius in feet.

On the basis of these values, the streamline tire requires only one-third as much solid friction as the airwheel. The shimmy tests, however, do not confirm this conclusion; the disagreement may be due to errors in the measured tire constants or the bump may have been more effective when the streamline tire was used. Considerable difficulty was encountered in determining $C_1 I_W$ and C_2, as is indicated by the range of values for T_{max}, 0.03 to $0.06\ r\mu W$ pound-feet, for the low-pressure tire.

Several factors investigated in these tests are considered in the theoretical analysis for both solid friction and viscous damping. In the theoretical analysis, the viscous damping was first developed and the solid-friction requirements were evolved by making reasonable assumptions. The solid friction required to prevent shimmy having been found to be adequately determined by

using measured constants in the equations of reference 1, it is logical to assume that this method provides equations adequate for calculating the viscous damping required.

Expressions for values of viscous damping, in which the measured constants in the equations developed by Kantrowitz (reference 1) have been substituted but in which the constant E_λ is treated as a constant times the square of the radius, are tabulated as follows:

Airwheel. . . . . . $K_{max} = 16\ r^2 \sqrt{I_w}\ \dfrac{\text{lb-ft}}{\text{radians per sec}}$

Low-pressure tire . $K_{max} = 26\ r^2 \sqrt{I_w}\ \dfrac{\text{lb-ft}}{\text{radians per sec}}$

Streamline tire . . $K_{max} = .32\ r^2 \sqrt{I_w}\ \dfrac{\text{lb-ft}}{\text{radians per sec}}$

These values are given only to indicate the range and the order of damping that may be expected and should not be considered as design values.

CONCLUSIONS

1. The amount of solid friction required to prevent shimmy was so great that it appears to be impracticable as the sole damping agent for castoring nose wheels on large airplanes.

2. The equation previously developed was found to be adequate for the prediction of solid friction. A value of T_{max} of 0.06 $r\mu W$ pound-feet (r is the tire radius in ft; μ, the coefficient of friction; and W, the load in lb) should provide sufficient solid friction for light airplanes.

3. The solid friction required to prevent shimmy:

(a) Decreased as the caster angle was moved from positive angles toward the vertical.

(b) Increased as the spindle moment of inertia was increased.

(c) Remained practically the same when the type or the pressure of the tire was changed.

Langley Memorial Aeronautical Laboratory,
National Advisory Committee for Aeronautics,
Langley Field, Va., April 1, 1940.

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2. Truscott, Starr: The N.A.C.A. Tank - A High-Speed Towing Basin for Testing Models of Seaplane Floats. T.R. No. 470, N.A.C.A., 1933.

3. Moyer, R. A.: Skidding Characteristics of Automobile Tires on Roadway Surfaces and Their Relation to Highway Safety. Engg. Exp. Sta. Bull. 120, Iowa State College, vol. XXXIII, no. 10, Aug. 8, 1934.

TABLE I

The Characteristics of the Five Tires Tested

Tire	Size (in.)	Over-all diameter (in.)	Rolling radius (in.)	Maximum allowable static load (lb)	Inflation pressure (lb/sq in.)
Airwheel (extra low pressure)	16 × 7-3	15.3	5.75	1,100	25
Low pressure	16 × 7-4	16.3	5.75	750	13
High pressure	16 × 4	15.8	7.1	750	65
Streamline	15.5	15.3	6.1	1,200	35
Solid	16 × 4	16.0	8.0	1,200	

TABLE II

Values of Tire Constants of Zero Caster Length

Tire	E_λ (lb/ft)	K_1 (ft^{-2})	$C_1 I_W$ (lb)	C_2 (ft^{-1})	Load (lb)
Airwheel	2,175	1.58	420	1.44	370
Low pressure	1,100	1.6	144	.49	225
Streamline	3,460	3.0	166	.51	320

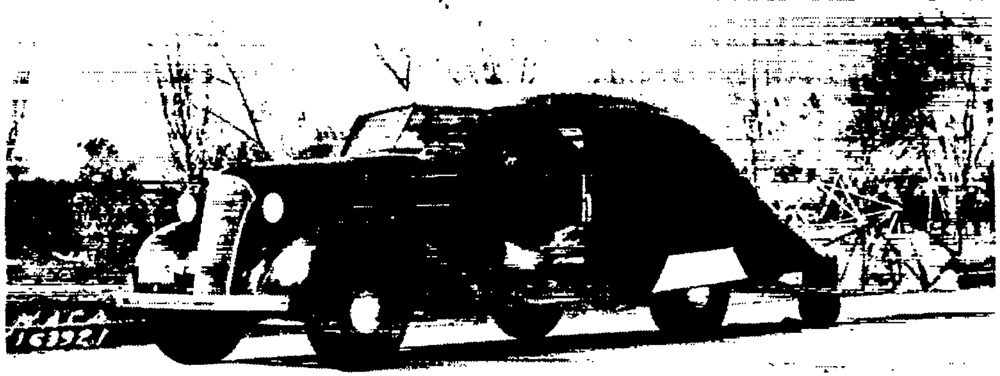

Figure 1.- Towcar with towcart attached. Castered wheel about to strike bump.

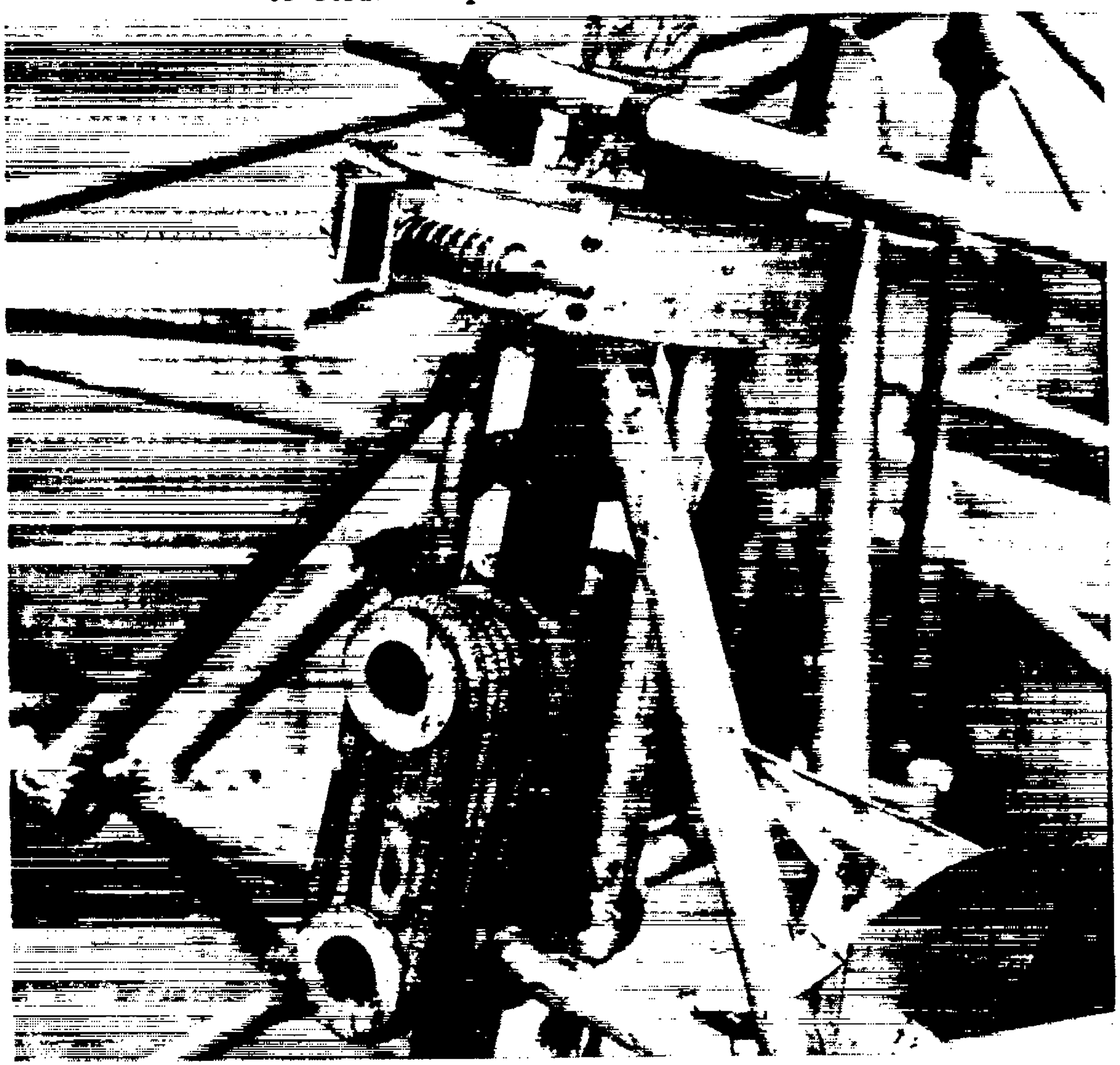

Figure 3.- Upper assembly of spindle and brake drum.

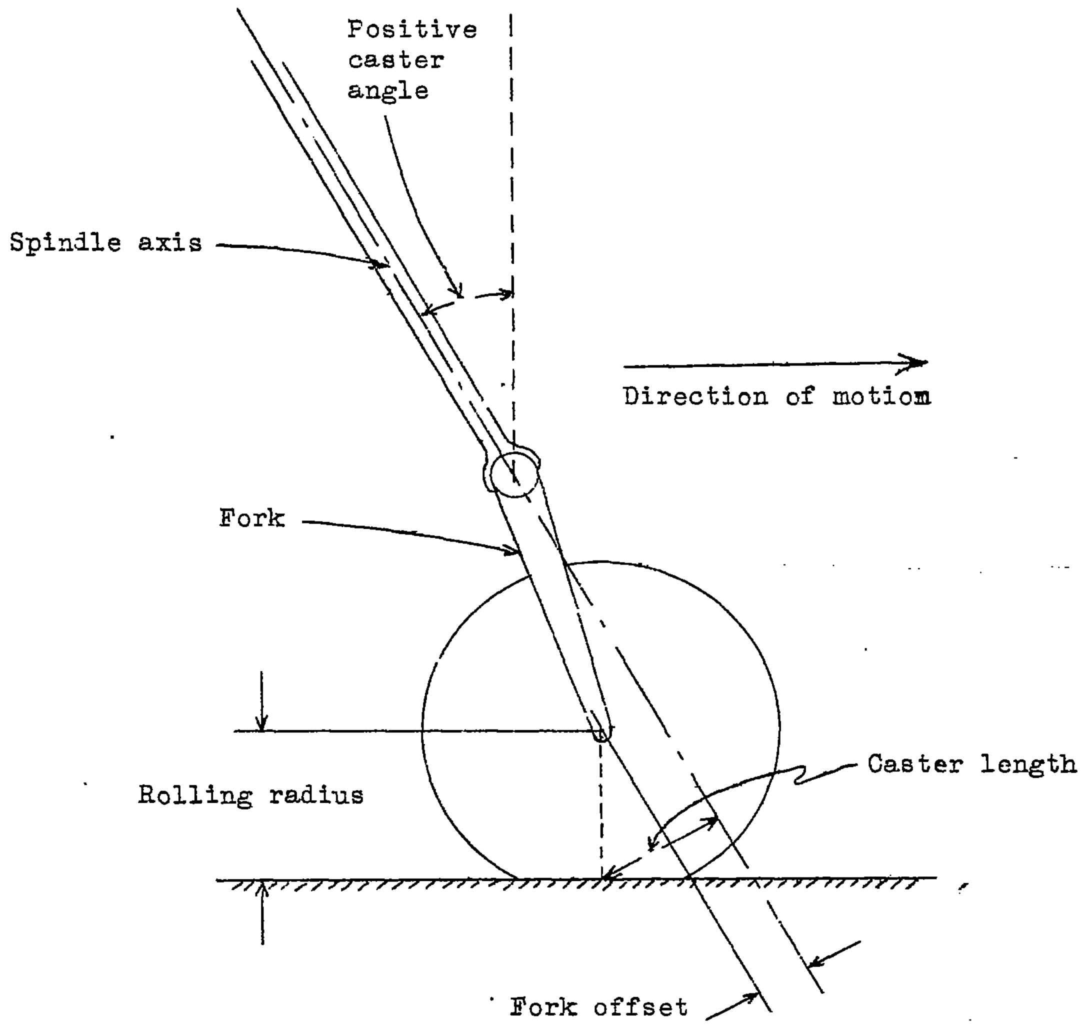

Figure 2.- Illustration of terms used.

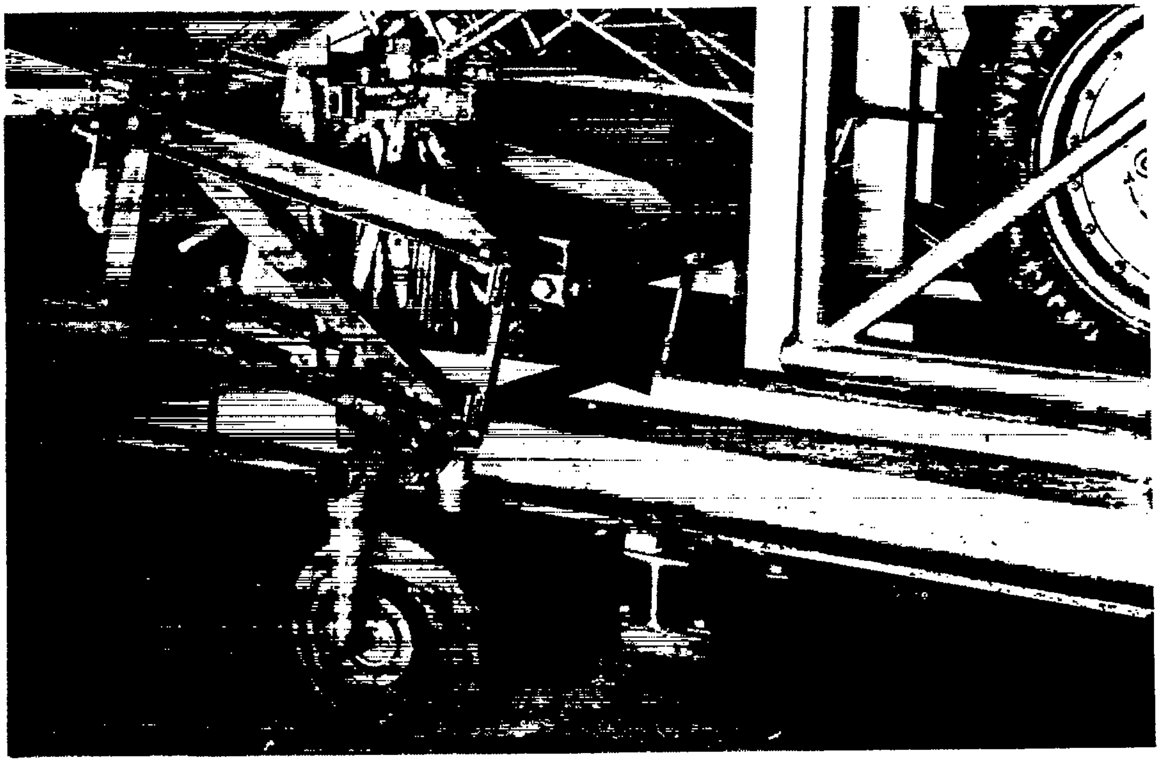

Figure 4.- Castering wheel and frame attached to N.A.C.A. tank towing carriage.

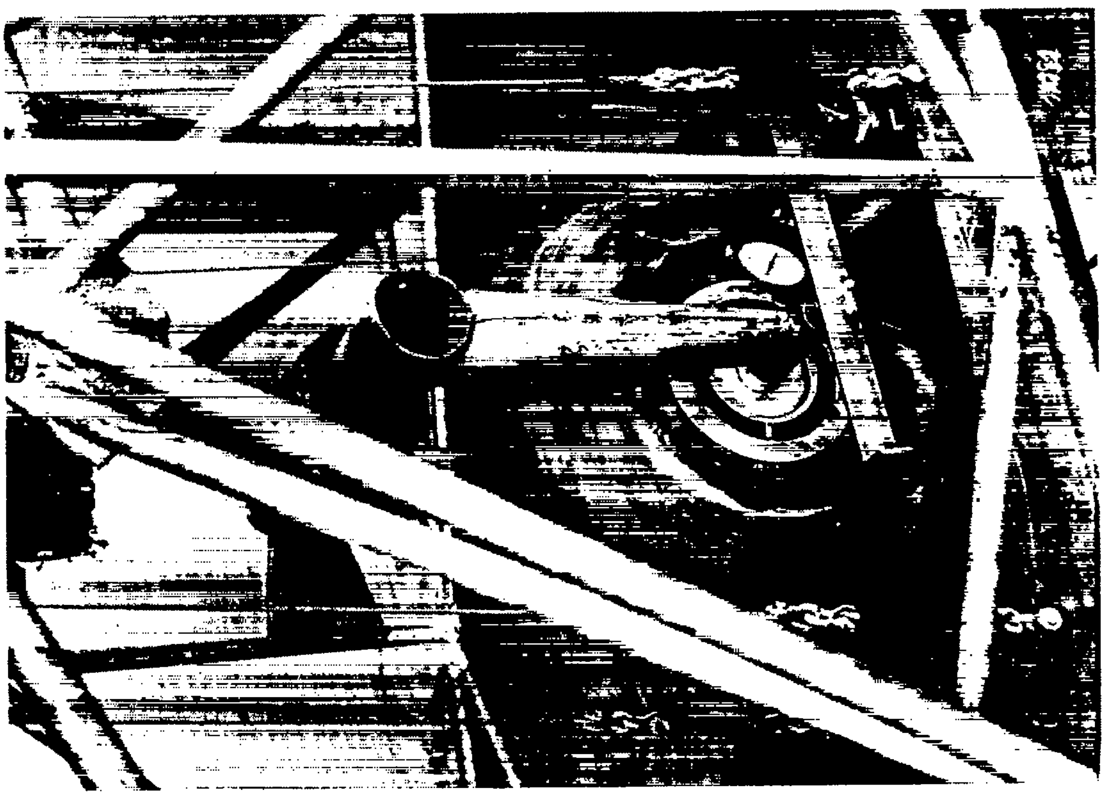

Figure 5.- Tire supported on a wire suspended platform with dial indicator in place.

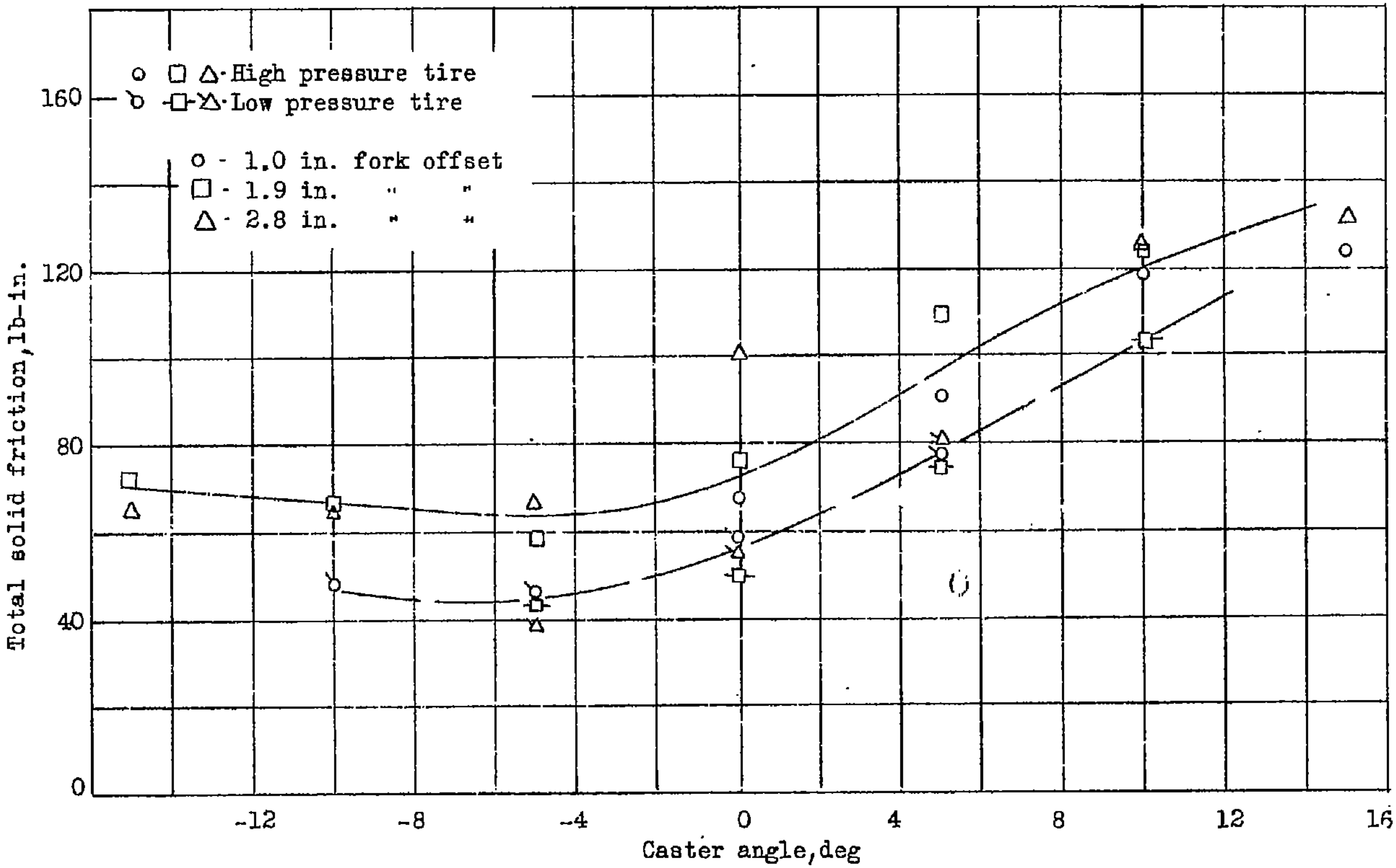

Figure 6.- Solid friction required to damp shimmy for low-and high-pressure tires. Load, 480 pounds; half-round bump, 1.5 inches high; V, 50 mph.

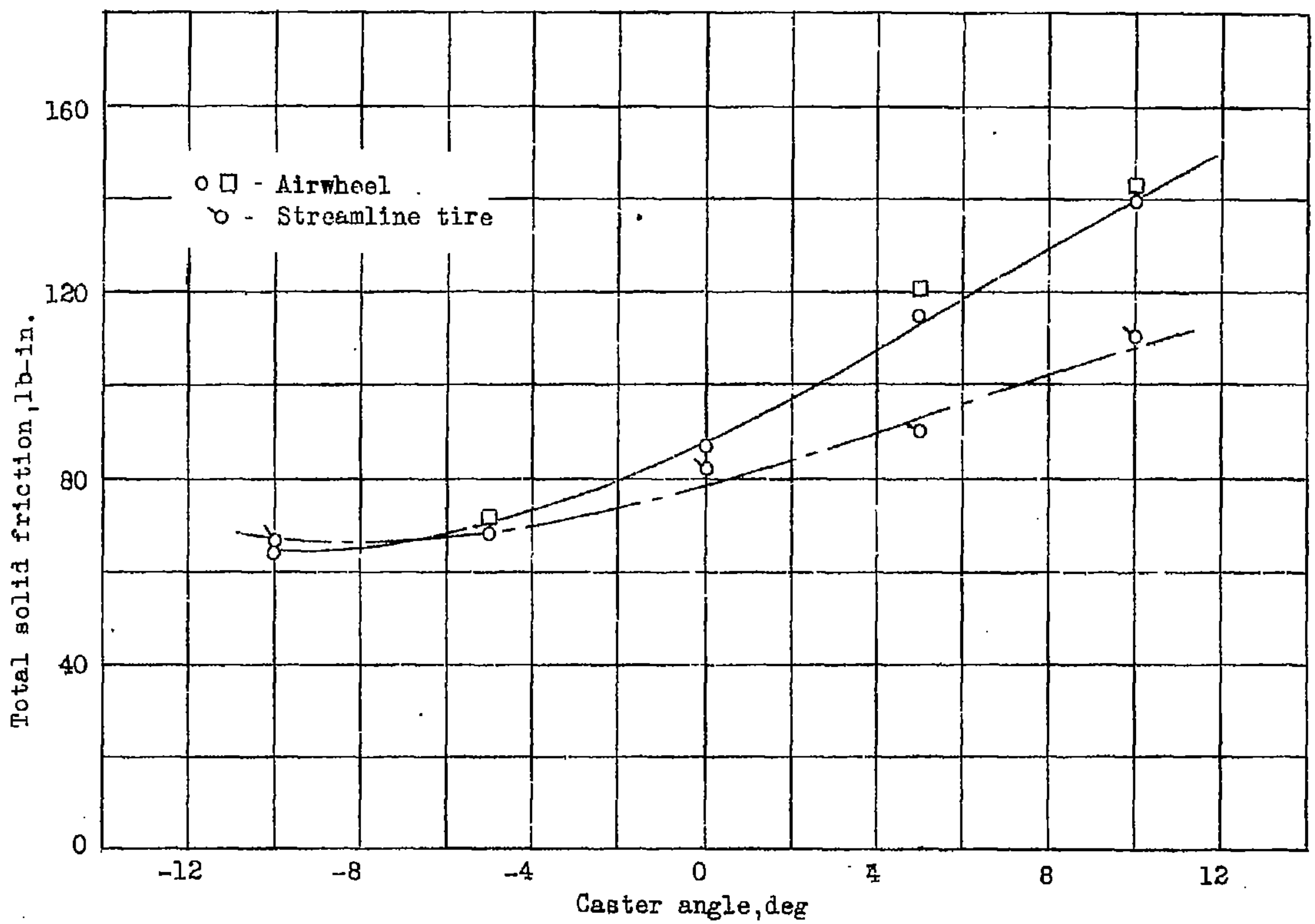

Figure 7.- Solid friction required to damp shimmy for airwheel and streamline tires. Load, 480 pounds; half-round bump, 1.5 inches high; V, 50 mph.

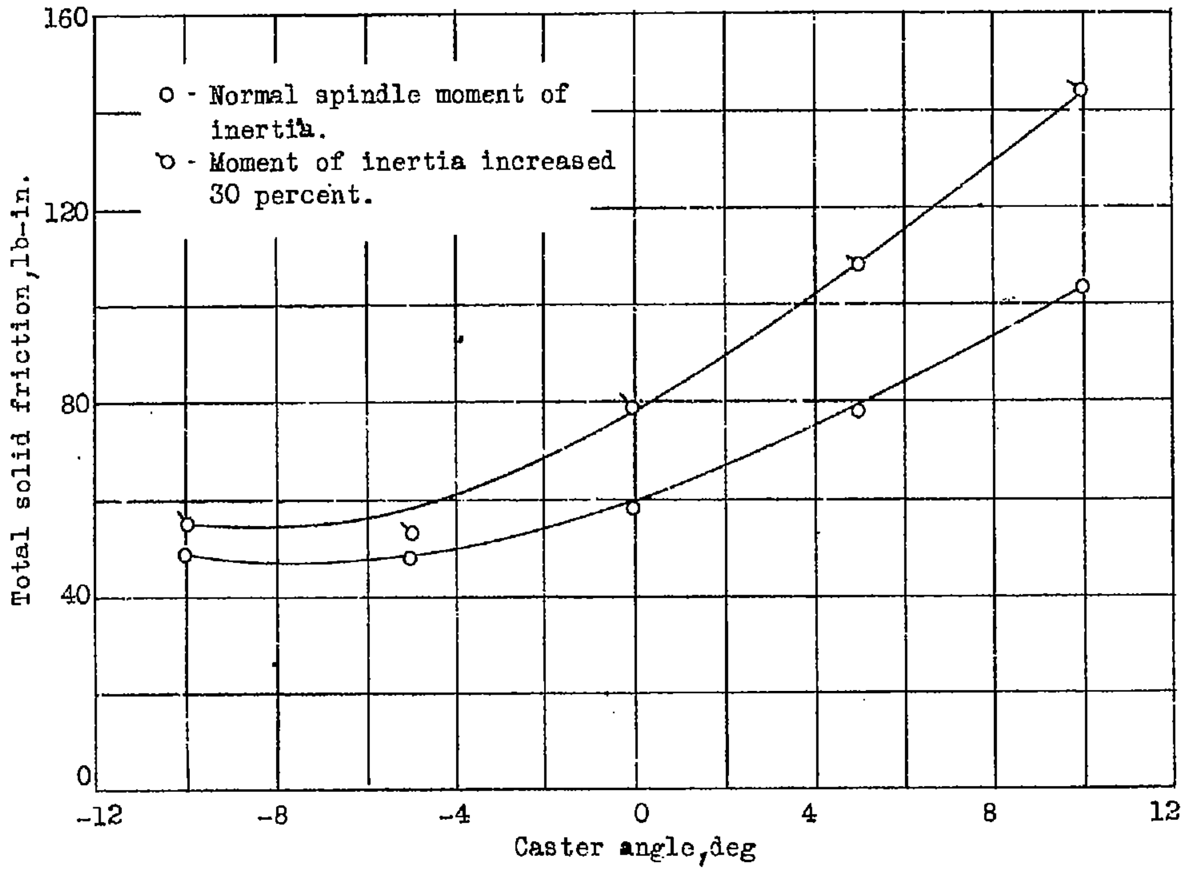

Figure 8.- Solid friction required to damp shimmy with variations in spindle moment of inertia. Low pressure tire; load,480 pounds; half-round bump,1.5 inches high; V, 50 mph.

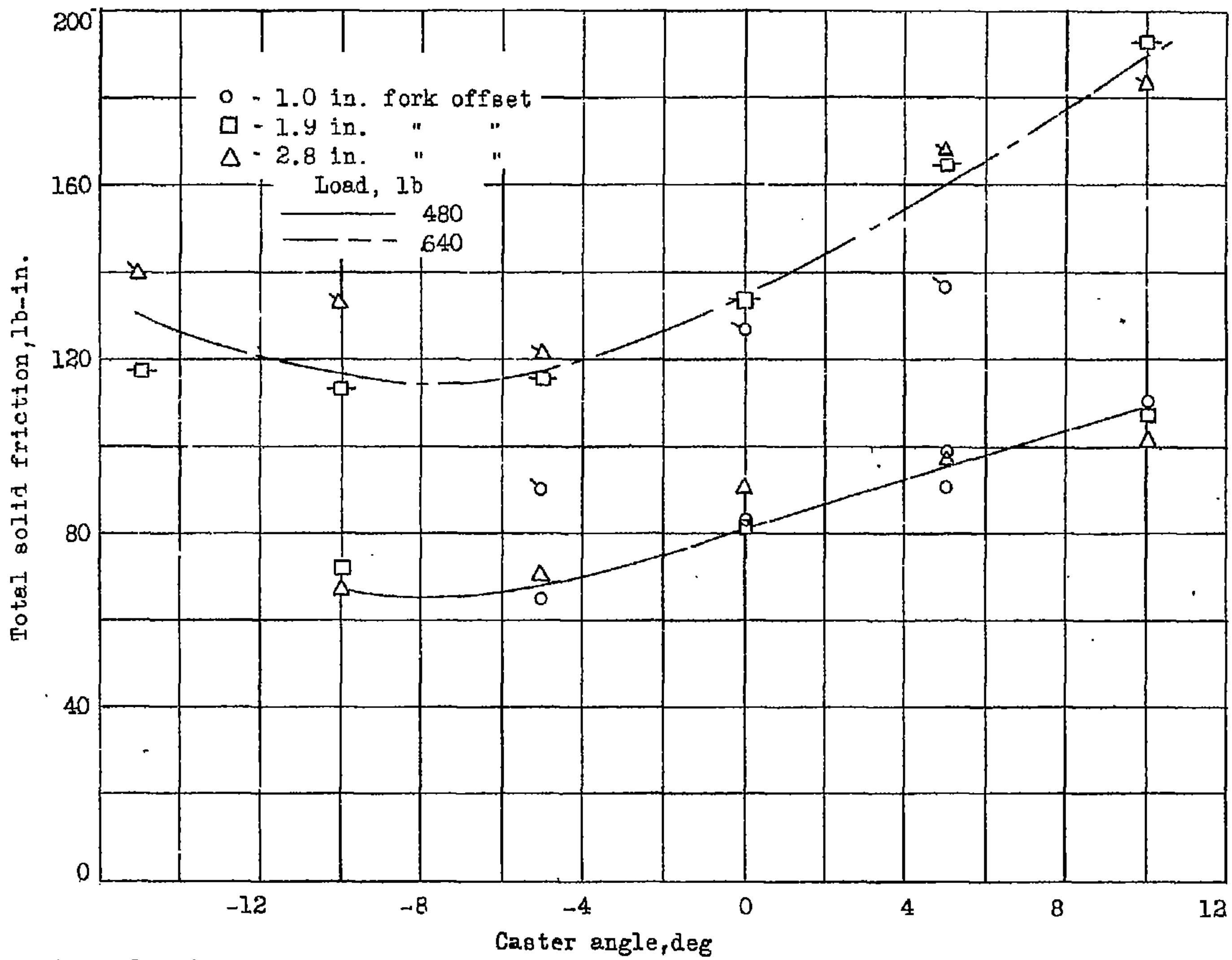

Figure 9.- Solid friction required to damp shimmy with variations in load and fork offset. Streamline tire; half-round bump, 1.5 inches high; V, 50 mph.

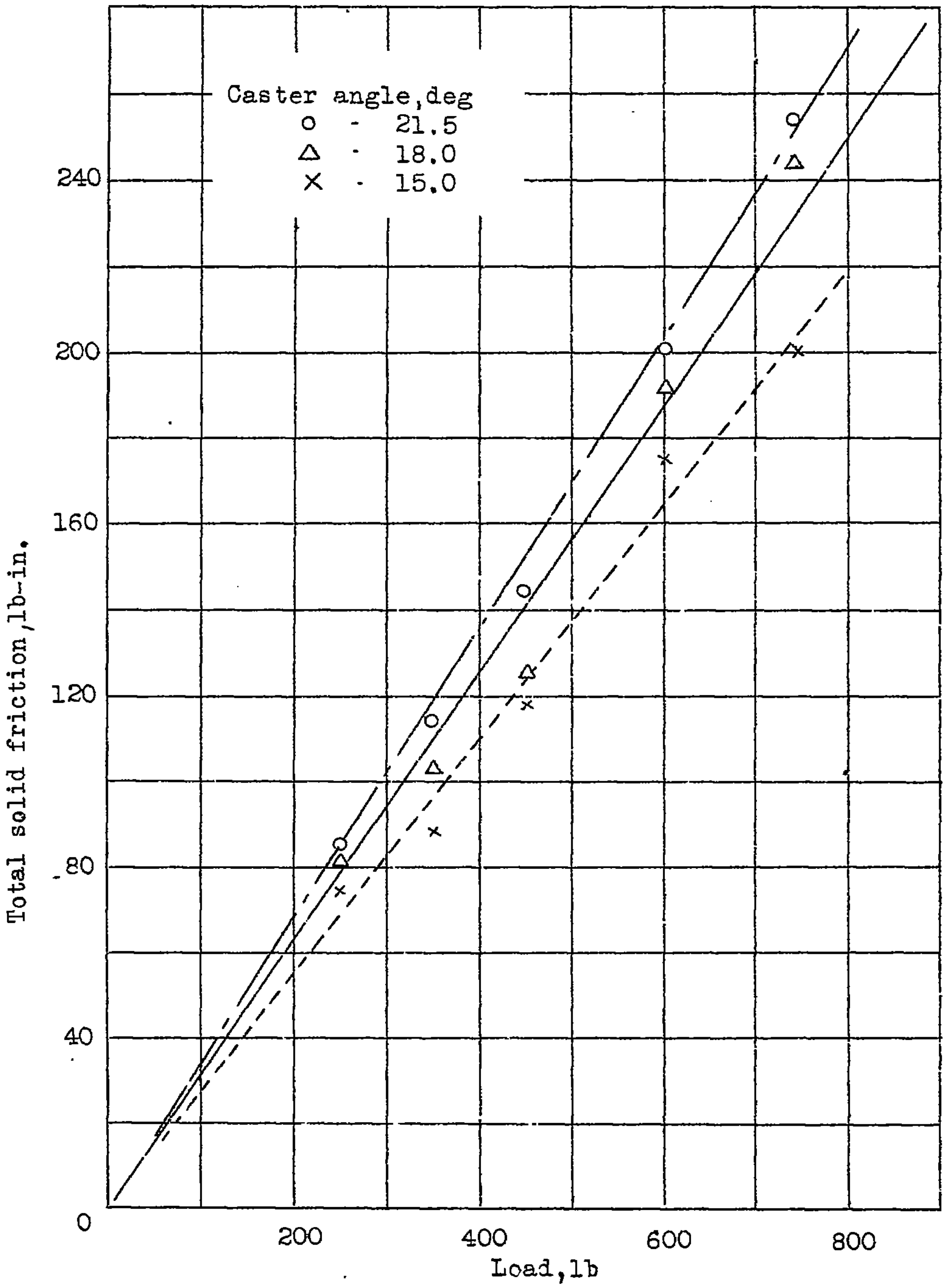

Figure 10.- Comparison of solid friction required at various loads and caster angles. Low pressure 16x7-4 tire;no fork offset,half-round bump,1.5 inches high; V, 50 mph.

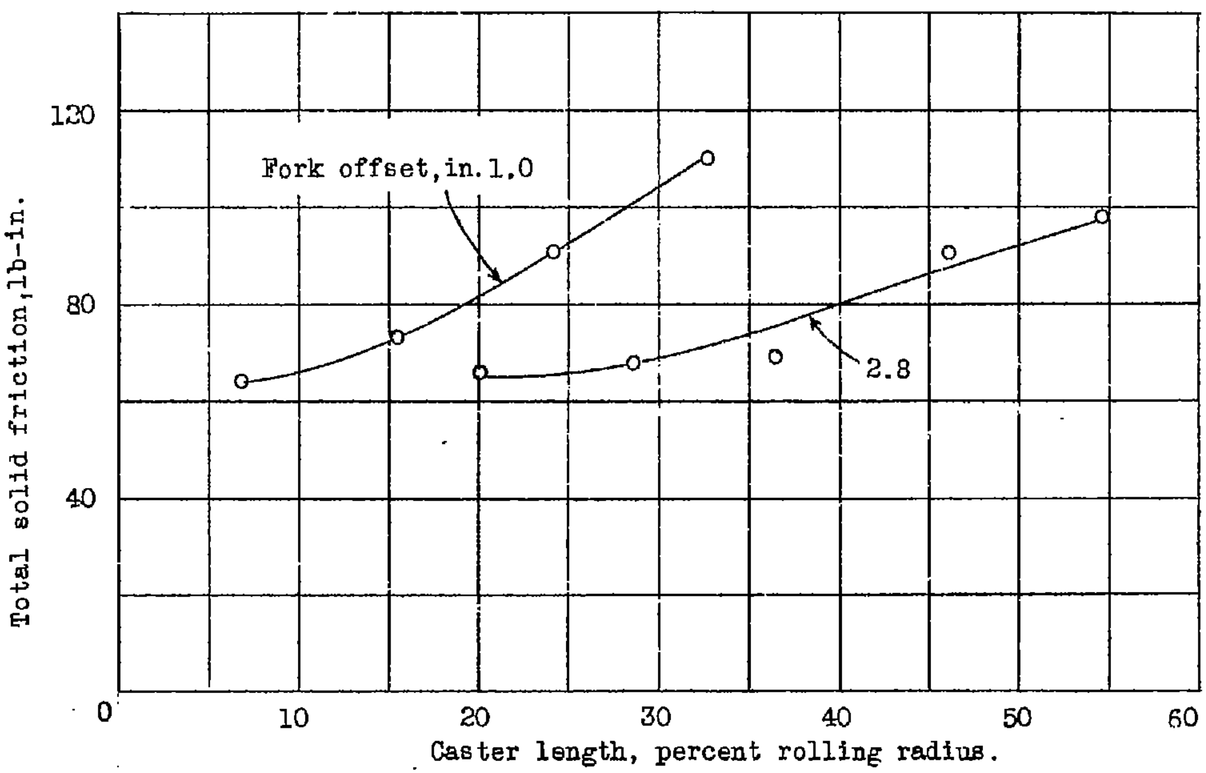

Figure 11.- Solid friction required to damp shimmy with variations in caster length and fork offset. Streamline tire; load, 480 pounds; half-round bump, 1.5 inches high; V, 50 mph.

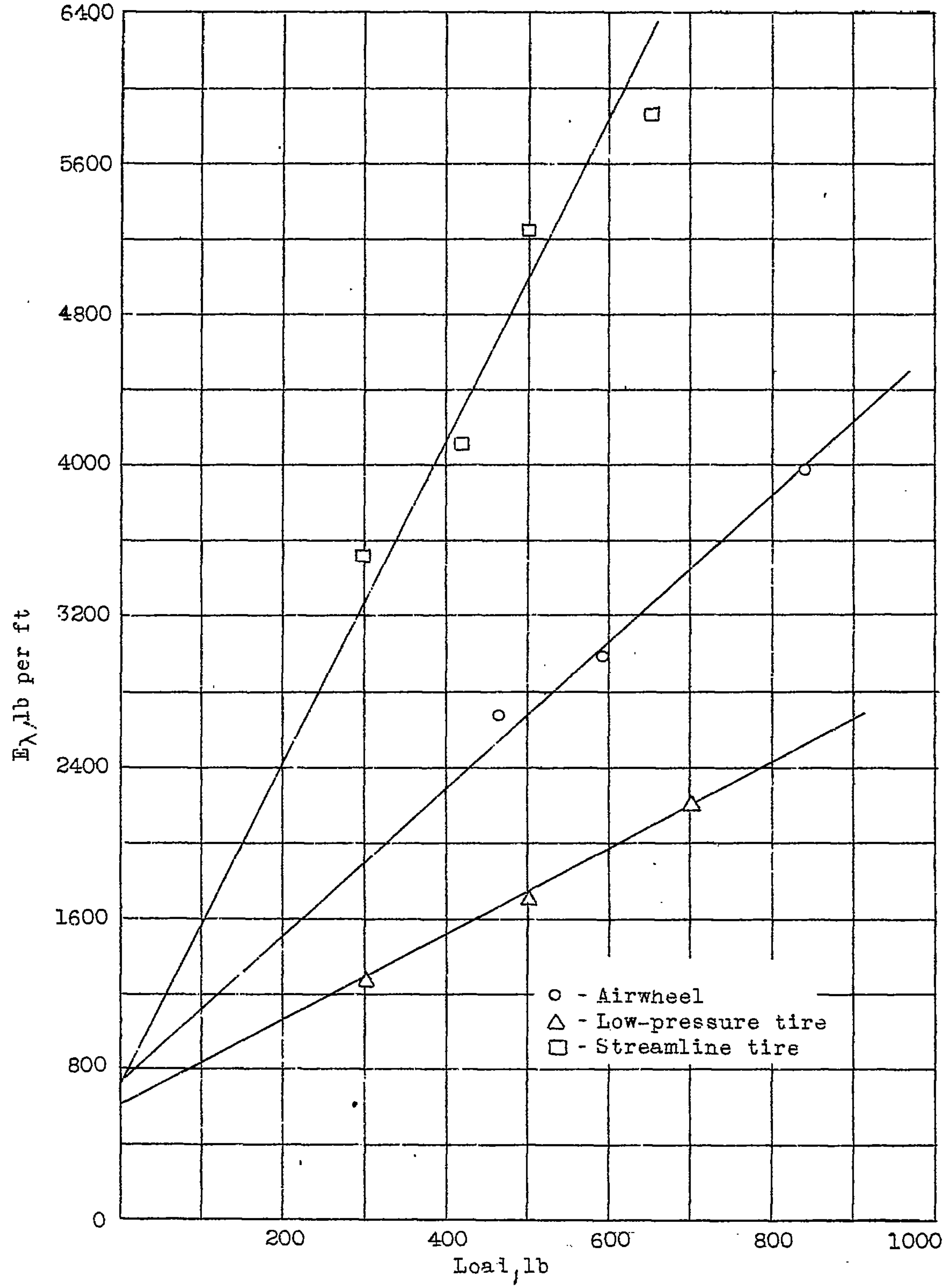

Figure 12.- Variation of E_λ with load.